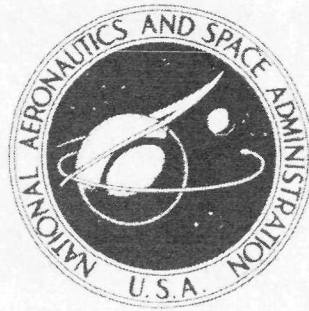


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EXPERIMENTAL STUDY OF LOW-SPEED
OPERATING CHARACTERISTICS OF A LIQUID
HYDROGEN CENTRIFUGAL TURBOPUMP

by Guy H. Ribble, Jr., and George E. Turney

Lewis Research Center

Cleveland, Ohio

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CENTRIFUGAL TURBOPUMP

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ABSTRACT

An experimental study was made of the low-speed operating characteristics of a liquid hydrogen centrifugal turbopump. The turbopump was operated at several speeds, ranging from 6.7 to 49 percent of the rated speed. The rated operating speed is 22 500 rpm or 2356 rad/sec. At the rated flow parameter of 0.346 gallon per minute per rpm ($2.09 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$), the pump isentropic efficiencies were 9, 32, and 63 percent for speeds of 1500, 3000, and 6000 rpm (157, 314, and 628 rad/sec), respectively. The isentropic efficiency was nearly constant for speeds above 6000 rpm (628 rad/sec). The head characteristics of the turbopump follow the affinity laws for the range of operating conditions investigated. At the pump rated flow parameter, an average total head parameter of $65 \times 10^{-6} \text{ ft}/(\text{rpm})^2$ ($18 \times 10^{-4} \text{ m}/(\text{rad/sec})^2$) was obtained.

EXPERIMENTAL STUDY OF LOW-SPEED OPERATING CHARACTERISTICS OF A LIQUID HYDROGEN CENTRIFUGAL TURBOPUMP

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SUMMARY

An experimental study was made of the low-speed operating characteristics of a liquid hydrogen centrifugal turbopump. The turbopump was operated at several speeds, ranging from 1500 to 11 000 rpm (157 to 1152 rad/sec). The rated operating speed is 22 500 rpm or 2356 radians per second.

At the rated flow parameter of 0.346 gallon per minute per rpm ($2.09 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$), the pump isentropic efficiencies were 9, 32, and 63 percent for speeds of 1500, 3000, and 6000 rpm (157, 314, and 628 rad/sec), respectively. The isentropic efficiency was nearly constant for speeds above 6000 rpm (628 rad/sec).

The head characteristics of the turbopump follow the affinity laws for the range of operating conditions investigated. At the pump rated flow parameter, an average total head parameter of 65×10^{-6} feet per $(\text{rpm})^2$ ($18.0 \times 10^{-4} \text{ m}/(\text{rad/sec})^2$) was obtained.

INTRODUCTION

The nuclear engine for rocket vehicle application (NERVA) program was established to develop a first generation nuclear rocket engine. In support of this development program, a nuclear rocket cold-flow test facility (designated the B-3 test facility) was constructed at the Plum Brook Station of Lewis Research Center. A detailed description of the B-3 test facility and of the full-scale, unfueled nuclear rocket engine is given in reference 1.

A series of experiments was conducted in the B-3 test facility to study and evaluate the performance of a full-scale nuclear rocket engine during startup transients. One facet of this experimental program dealt with use of a bootstrap technique to power the propellant turbopump during engine startup. The time period for startup of a nuclear rocket is relatively long (approx. 30 sec) compared to that of a chemical rocket.

Throughout a large part of the nuclear rocket startup, the propellant turbopump operates in a range of relatively low speeds. A knowledge of the performance of the turbopump at these low operating speeds is necessary to select a successful operating schedule for bootstrap startups of a nuclear rocket.

In order to investigate the turbopump performance in the range of low-speed operation, a series of low-speed turbopump tests was run in the B-3 test facility. The results of these low-speed turbopump tests are presented and discussed in this report.

Normally, pump capacity, pump head, and brake horsepower vary with speed according to the affinity laws. These laws state that (1) pump capacity varies directly as the speed, (2) pump head varies directly as the square of the speed, and (3) brake horsepower varies directly as the cube of the speed. The horsepower relation is based on the assumption that pump efficiency remains constant as speed varies at a constant specific speed.

In the low-speed range of operation, the pump affinity laws may not apply. This is because certain losses which are insignificant at normal operating speeds become predominate at low speeds. There appear to be no reliable analytical methods available for predicting pump behavior in the low-speed region. In a bootstrap startup of the nuclear rocket, the turbopump must operate through a range of low speeds. Hence, the performance of the turbopump in the low-speed region is of special interest.

The turbopump tested and described in this report was manufactured by the Aerojet General Corporation. This turbopump, designated the Mark III - Model 4, consists of a single-stage centrifugal pump driven by a two-stage turbine. A turbopump of this type has been used in ground demonstration tests of the NERVA engine.

Performance data for this pump were provided by the manufacturer. However, the manufacturer's data did not include the range of low-speed operation. The pump tests conducted at Lewis and reported herein cover a speed range from 6.7 to 49.0 percent of rated speed. The rated speed of the Mark III - Model 4 turbopump is 22 500 rpm (2356 rad/sec).

SYMBOLS

g	local gravitational acceleration, approx. 32.174 ft/sec^2 ; 9.80 m/sec^2
g_c	conversion factor, $32.174 \text{ (lbm)(ft)/(lbf)(sec}^2\text{)}$; $1.0 \text{ (kg)(m)/(N)(sec}^2\text{)}$
ΔH	head rise across pump, feet of liquid hydrogen; meters of liquid hydrogen
h	enthalpy, Btu/lbm; J/g
K	constant (eq. (6)), $144 \text{ in.}^2/\text{ft}^2$; $1 \times 10^{-3} \text{ (g)(m}^3\text{)/(kg)(cm}^3\text{)}$
N	pump rotational speed, rpm; rad/sec

P	pressure, psia; N/m^2
Q_D	volumetric flow rate, gal/min; m^3/sec
S	entropy, $\text{Btu}/(\text{lbm})(^\circ\text{R})$; $\text{J}/(\text{g})(\text{K})$
T	fluid temperature, $^\circ\text{R}$; K
η	isentropic efficiency
ρ	hydrogen density, lbm/ft^3 ; g/cm^3

Subscripts:

D	discharge (or outlet) side of pump
i	ideal, see eq. (1)
S	suction (or inlet) side of pump

APPARATUS AND EXPERIMENTAL PROCEDURE

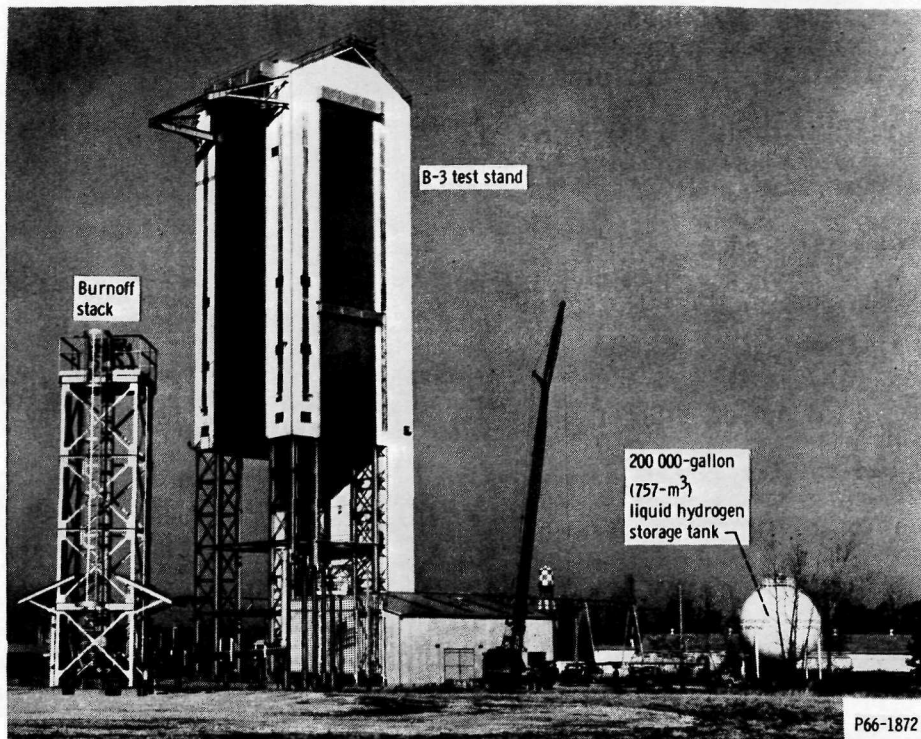
Research System

The B-3 test facility is shown in figure 1. The research system (i. e., the full-scale, unfueled nuclear rocket engine) is located inside the B-3 test stand. It consists of a 40 000-gallon (151.4-m^3) propellant storage tank, reactor, nozzle, turbopump and piping. The assembled configuration of the research system is shown in figure 2. A detailed description of the major components of this system is given in reference 1.

Information obtained from studies conducted with this research system has been reported. References 2 and 3, for example, give comparisons of the experimental and predicted pressure-drop-heat-transfer characteristics of the reactor core and nozzle assembly, respectively. And reference 4 describes an analytical model, developed from test data, for predicting the low frequency dynamics of the nuclear rocket engine.

The research system depicted in figure 2 was modified somewhat for low-speed turbopump experiments. For these tests, the turbopump was disconnected from the propulsion nozzle, and independent turbine supply and pump discharge lines were connected to the turbopump. Figure 3 is a schematic of the test configuration used for the low-speed turbopump testing.

Gaseous hydrogen to power the turbine was supplied from a 2400-psig ($1.65 \times 10^7 \text{ N/m}^2$ -gage) tank farm. The high pressure supply gas was regulated and reduced to approximately 100 psia ($6.89 \times 10^5 \text{ N/m}^2$) at the turbine inlet. The gaseous hydrogen flow was measured with a calibrated venturi meter. The rotational speed of the turbine was regulated by the turbine power control valve (TPCV) located just ahead of the turbine in-



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Figure 1. - B-3 test facility.

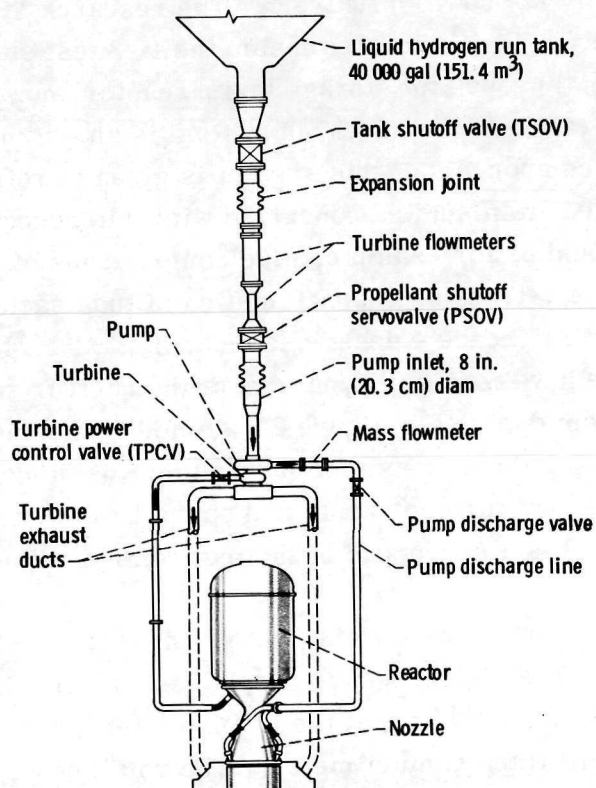
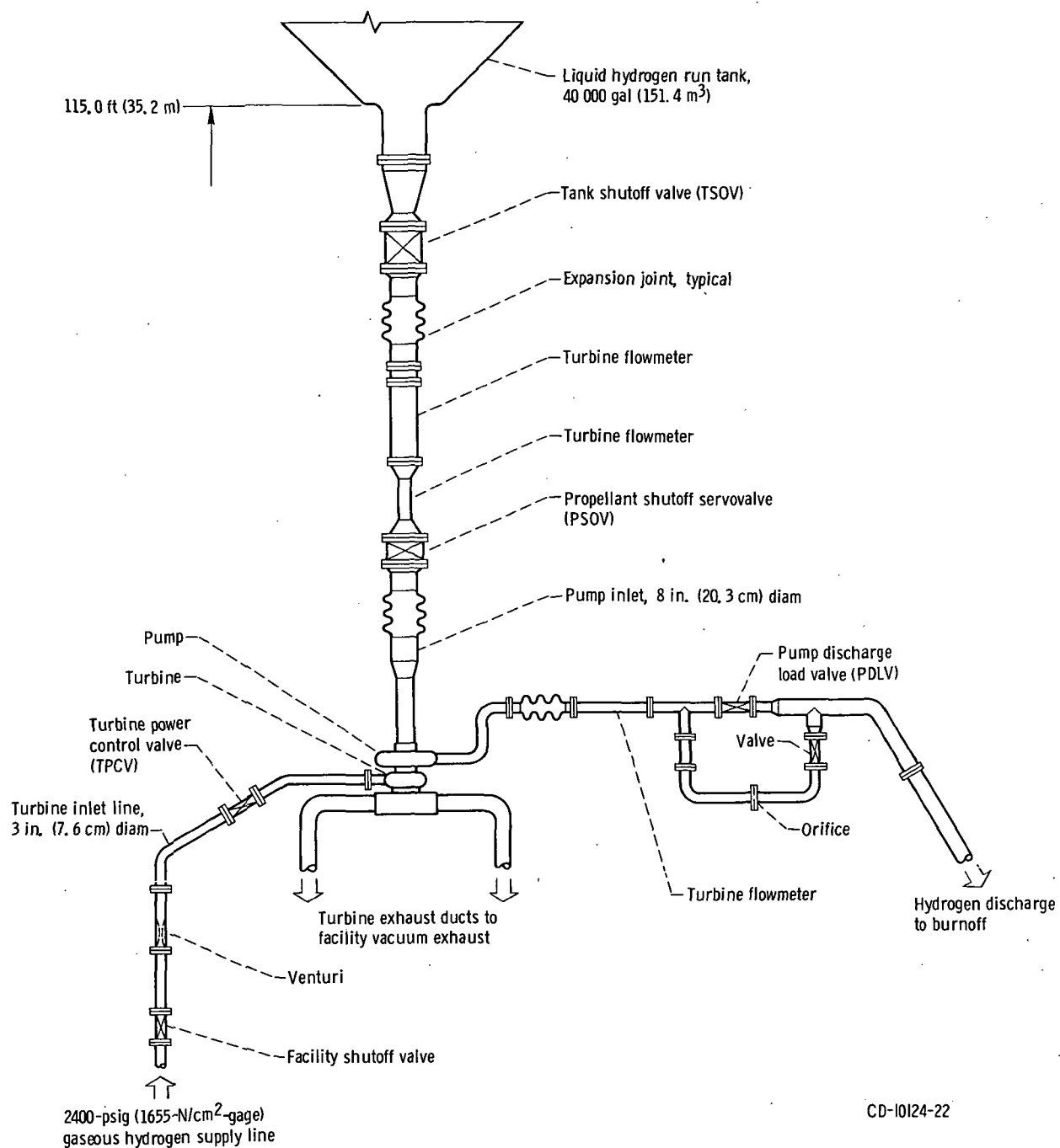


Figure 2. - Full-scale, cold-flow research system.



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Figure 3. - System configuration for Mark III - Model 4 turbopump tests.

let. The exhaust gas from the turbine was transferred through two parallel exhaust lines to a facility ejector.

The liquid hydrogen flow rate was measured at three places in the system. Two turbine flowmeters were located in series in the pump inlet line and one turbine flowmeter was located in the pump discharge line. The turbopump and pump inlet and discharge lines were insulated with a 4-inch (10.16-cm) thick covering of polystyrene foam in the form of small spheres about 1/8 inch (0.318 cm) in diameter. The insulation was purged with ambient temperature helium before tests to keep air and moisture from condensing on the lines during a test run.

As indicated in figure 3, the pump discharge flow system contained a bypass around the pump discharge load valve (PDLV). The bypass contained a valve and orifice. The pump discharge flow from the bypass section was transferred through a 6-inch (15.24-cm) nominal diameter pipe line to a facility exhaust stack where the hydrogen was burned.

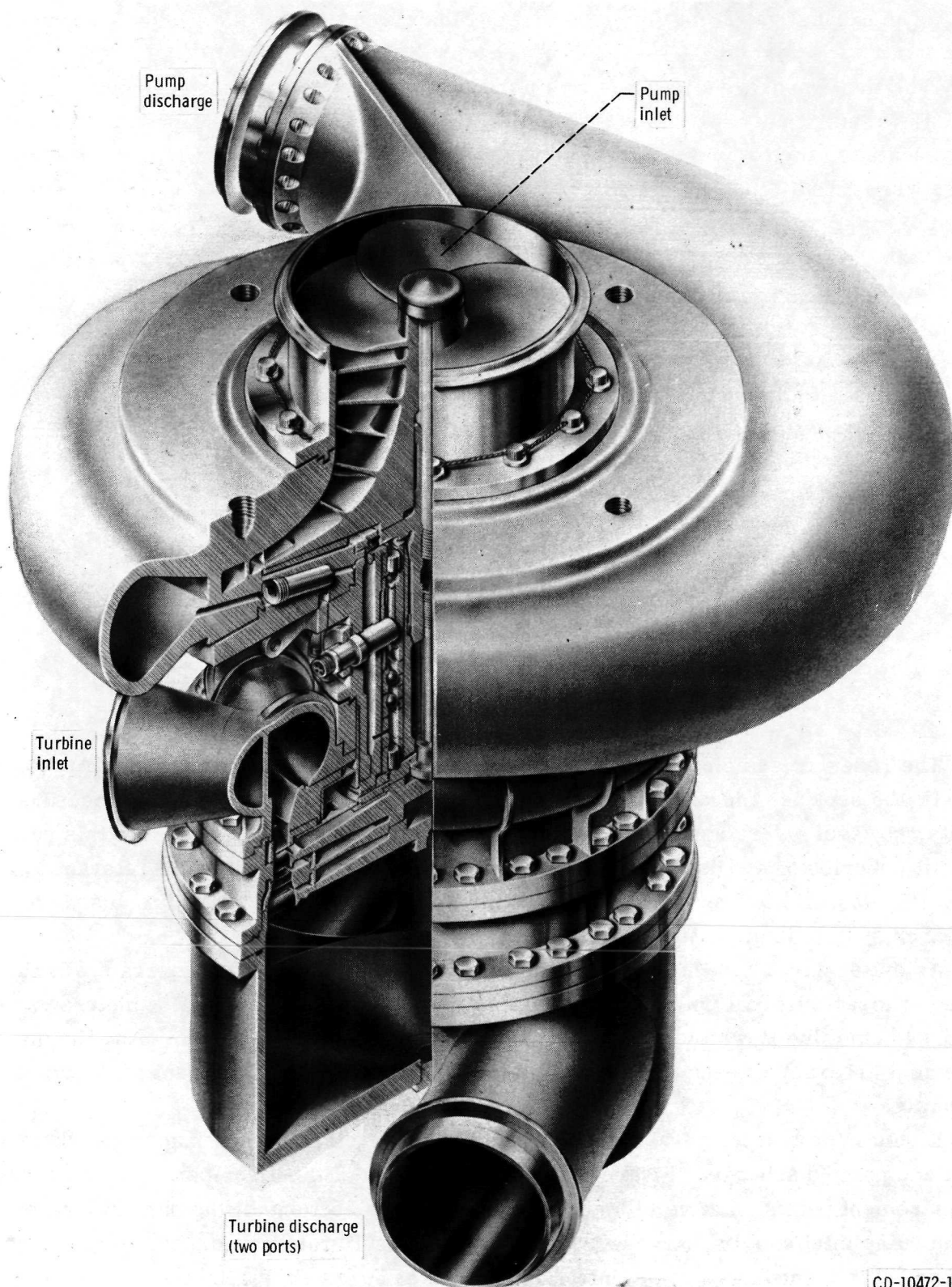
Turbopump

The Mark III - Model 4 turbopump assembly is shown in figure 4. The pump impeller and turbine wheels of this assembly are cantilevered from opposite ends of a common shaft. The shaft is supported in the center by two roller bearings and two ball bearings.

The pump rotor inlet has a 7° (0.122-rad) helical inducer with a tip diameter of 6.95 inches (17.65 cm) and a solidity of 1.85 at the tip. The inducer is integral with the centrifugal impeller. Figure 5 shows the pump inducer-impeller combination. The centrifugal impeller has several 0.55-inch (1.40-cm) high radial discharge blades and a diameter of 12.50 inches (31.75 cm) at the discharge end.

The pump casing has a double volute with 0.55-inch (1.40-cm) high diffuser vanes set at a 9.25° (0.1614-rad) angle. The volute discharge nozzle has an inside diameter of 4.62 inches (11.73 cm). The rated conditions for this turbopump are 1000-psi (6.895×10^6 N/m²) pressure rise at a liquid hydrogen flow rate of 76 pounds mass per second (34.5 kg/sec). This rating is based on a net positive suction head of 10 psi (6.89×10^4 N/m²). Rated speed is 22 500 rpm (2356 rad/sec).

The turbine is a two-stage, pressure-compounded impulse type. The mechanical bearings of the turbopump are lubricated and cooled with liquid hydrogen bled from the pump discharge. The coolant from the bearings flows into the turbine and mixes with the main stream turbine gas. At rated conditions, the flow rate through the bearings is about 0.3 percent of the pump flow rate.



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Figure 4. - Mark III, Model 4 turbopump.

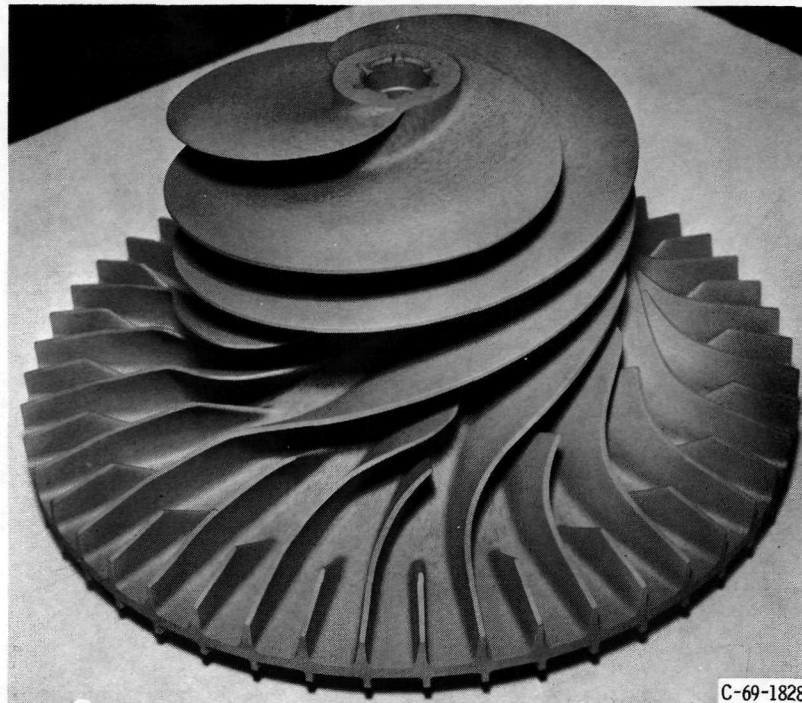


Figure 5. - Mark III - Model 4 inducer and impeller.

Instrumentation and Accuracy

The research system depicted in figure 3 was extensively instrumented. Approximately 125 separate measurements were made on this system. Most of the measurements, however, were used for test monitoring and diagnostic purposes. In this section, we will describe only those items of instrumentation which were directly relevant to the Mark III - Model 4 turbopump tests. A complete list of the instrumentation used on the research system is given in reference 1.

As stated earlier in this section, the liquid hydrogen turbopump flow rate was measured at three different points in the system. Precision-type turbine flowmeters were used. Each of the flowmeters (see fig. 3) was independently calibrated. And the measurement accuracy of each was estimated to be within about ± 1.5 percent of the actual flow rate.

Liquid hydrogen temperature and pressure measurements were made near the pump inlet and pump discharge. Figure 6 shows the location of these temperature and pressure measurement points relative to the inlet and discharge sections of the pump. Pressures at the pump inlet and discharge were measured with calibrated strain-gage-type transducers. The transducers at the inlet had a range of 50 psi (3.45×10^5 N/m²). At the

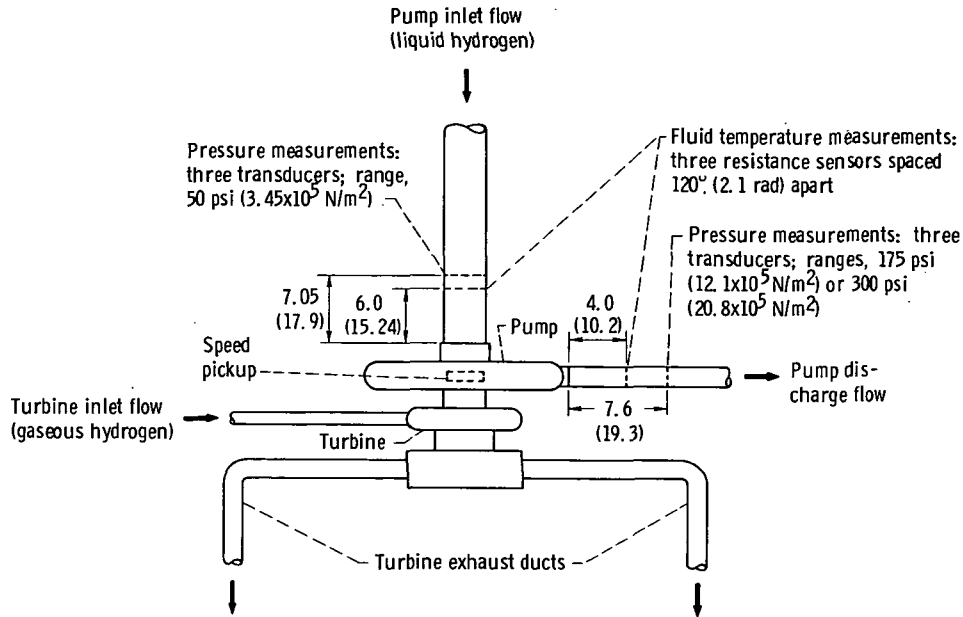


Figure 6. - Mark III - Model 4 turbopump and associated instrumentation. Dimensions are in inches (cm).

pump discharge section, two different transducer ranges were used. For one of the tests, a range of 175 psi (12.1×10^5 N/m²) was used. For the other tests, the range was 300 psi (20.7×10^5 N/m²).

In addition to the inlet and outlet pressure transducers, differential pressure transducers with ranges from 50 to 300 psi (3.45×10^5 to 20.7×10^5 N/m²) were used to measure pump pressure rise. However, due to the large pressure rise across the pump, the differential transducers offered little improvement in the measurement accuracy of pump pressure rise.

The precision of the measured pressures was estimated to be within $\pm 1/2$ percent of the full-scale transducer range. Thus, pump inlet pressure measurements were accurate to within ± 0.5 psi ($\pm 3.45 \times 10^3$ N/m²). And measurements at the pump outlet were accurate to within ± 0.825 psi ($\pm 5.69 \times 10^3$ N/m²) or ± 1.50 psi ($\pm 10.34 \times 10^3$ N/m²), depending on the range of the transducers.

Liquid hydrogen temperatures were measured with precision platinum resistance sensors. The locations of the sensors are indicated in figure 6. The three sensors in the inlet line were spaced 120° (2.1 rad) apart. They extended radially into the stream so that the distance from the pipe centerline to the sensing junction was equal to 72 percent of the pipe inside radius. The three sensors in the discharge line were similarly arranged.

The platinum-resistance temperature sensors were independently calibrated after the test runs. Based on the calibration data, corrections were applied to the measured tem-

peratures. The precision of the corrected measurements was determined to be within about $\pm 0.08^\circ \text{R}$ ($\pm 0.0444 \text{ K}$).

The rotational speed of the turbopump was measured by two electromagnetic pickups mounted inside the pump case. The pulse frequencies from these pickups were converted to a direct-current signal, proportional to the pump speed. The accuracy of the speed measurement was estimated to be within ± 1 percent.

Test Procedure

The Mark III - Model 4 turbopump was tested at five different rotational speeds, ranging from 1500 to 11 000 rpm (157 to 1152 rad/sec). Propellant tank pressures of 35 and 50 psia (2.42×10^5 and $3.45 \times 10^5 \text{ N/m}^2$) were maintained for each of the five pump speeds. A summary of the test runs is given in table I.

As indicated in table I, the conditions of test 3 were the same as those for test 2. The main purpose of test 3 was to determine the reproducibility of the test data.

TABLE I. - SUMMARY OF MARK III -
MODEL 4 TURBOPUMP TESTS

Test	Rotational speed		Tank pressure	
	rpm	rad/sec	psia	N/m^2
1	1 500	157	35	2.42×10^5
	3 000	314	35	2.42
	6 000	628	35	2.42
	1 500	157	50	3.45
	3 000	314	50	3.45
	6 000	628	50	3.45
2	6 000	628	35	2.42×10^5
	9 000	942	35	2.42
	11 000	1152	35	2.42
	6 000	628	50	3.45
	9 000	942	50	3.45
	11 000	1152	50	3.45
3	6 000	628	35	2.42×10^5
	9 000	942	35	2.42
	11 000	1152	35	2.42
	6 000	628	50	3.45
	9 000	942	50	3.45
	11 000	1152	50	3.45

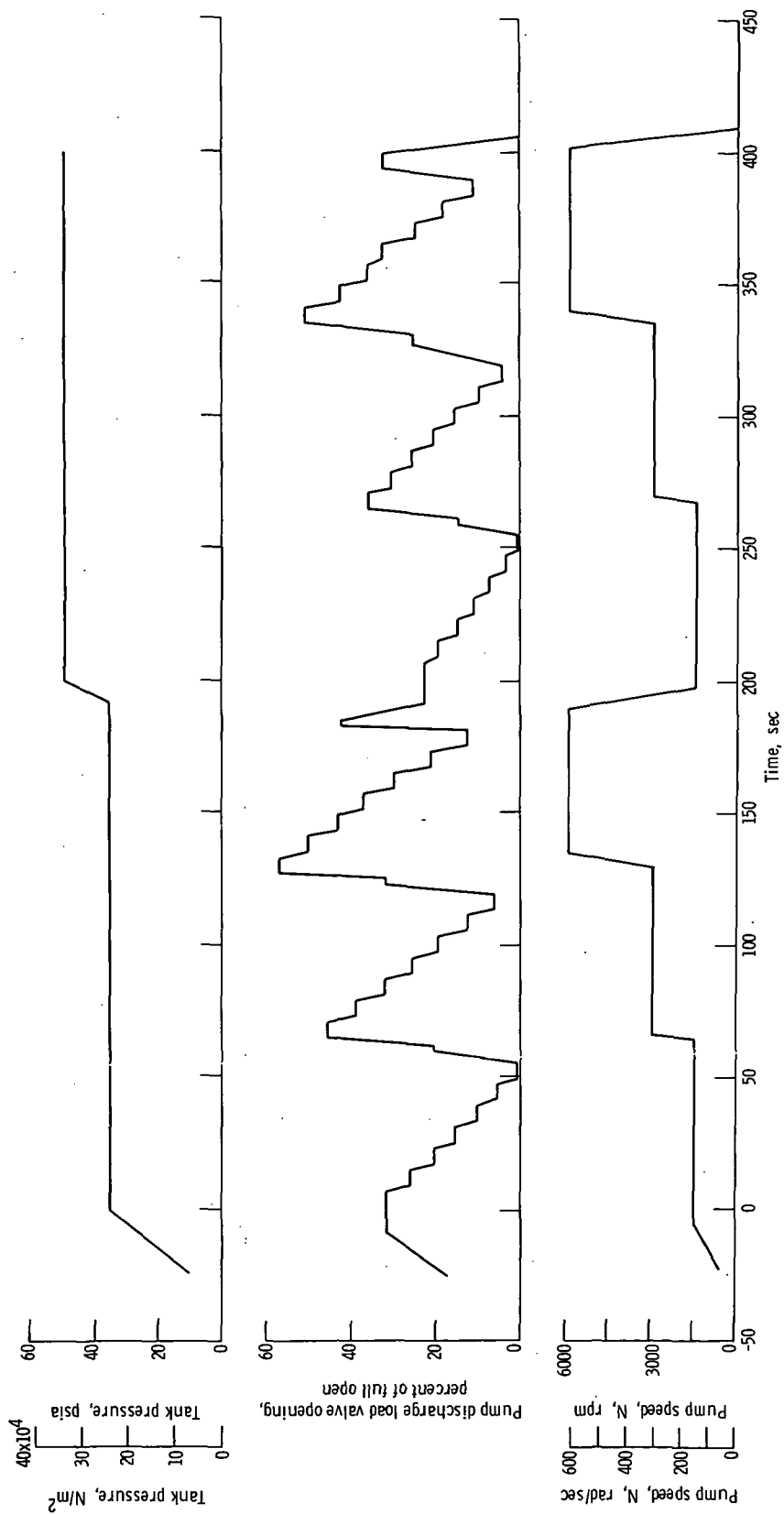


Figure 7. - Simplified illustration of sequence of events for turbopump test 1.

The procedure used in the operation of the turbopump test runs is fully described in reference 1. The following is a summary of this procedure along with a description of the events taking place during a test run.

Prior to the start of the pump experiments, the facility is checked out and prepared for operation. The prerun procedure includes electronic calibrations of instrumentation, evacuation and helium purge of the entire system, and filling of the liquid hydrogen storage (run) tank. Next, the turbopump and pump inlet and discharge lines are cooled to approximately the temperature of liquid hydrogen. The cooldown is accomplished by a continuous flow of liquid hydrogen through the system. The temperatures of the turbopump and piping were carefully monitored during the chilldown. The liquid hydrogen flow rate for chilldown was about 4 pounds mass per second (1.81 kg/sec). To ensure a complete chilldown, this flow rate was maintained for a period of approximately 30 minutes.

The three tests summarized in table I were run in a similar fashion. A simplified sequence of events for test 1 is depicted in figure 7. First, the pressure in the liquid hydrogen storage tank is increased to 35 psia ($2.42 \times 10^5 \text{ N/m}^2$). Then, the pump discharge load valve (PDLV) is opened to a position which gives the maximum desired flow rate. Turbopump speed is ramped to 1500 rpm (157 rad/sec) by opening the turbine power control valve (TPCV) to a set position.

While maintaining a speed of 1500 rpm (157 rad/sec), the PDLV is closed in six steps. Sufficient time is allowed at each step for steady-state conditions to be reached. Data were recorded at each of the seven PDLV settings. An orifice in the PDLV bypass line allows the turbopump to enter stall, but prevents the flow from decreasing to a level where pump damage may result.

The PDLV is then ramped at a rate of 10 percent of stroke per second to the maximum desired test position. Data were recorded during the early part of this ramp. The pump is accelerated to 3000 rpm (314 rad/sec), and then to 6000 rpm (628 rad/sec), repeating the above procedure at each speed. At the end of the 6000-rpm (628-rad/sec) test run, the PDLV is opened to an intermediate position where the pump is out of stall. And the pump speed is decreased at a constant rate to 1500 rpm (157 rad/sec). The tank pressure is then increased to 50 psia ($3.45 \times 10^5 \text{ N/m}^2$), and the runs at 1500, 3000, and 6000 rpm (157, 314, and 628 rad/sec) are repeated. Following the 6000-rpm (628-rad/sec) run, the pump is decelerated to zero.

Data Acquisition and Processing

All measurements pertinent to the pump tests were recorded on a digital multiplex system. Data for each channel were recorded at a frequency of 100 readings per second.

The raw test data were processed on a digital computer which converted the respective millivolt signals to engineering units of temperature, pressure, flow rate, and pump

speed. A 10-sample group averaging technique was used in the data processing. And the converted experimental measurements were printed out by the computer at a frequency of 10 values per second.

Data Analysis

Turbopump efficiency. - The turbopump isentropic efficiency was determined from the following equation:

$$\eta = \frac{\text{Ideal enthalpy change}}{\text{Actual enthalpy change}} = \frac{h_{D,i} - h_S}{h_D - h_S} \quad (1)$$

The ideal discharge enthalpy $h_{D,i}$ is a function of the pump discharge pressure P_D and the pump inlet entropy S_S ; that is,

$$h_{D,i} = f(P_D, S_S) \quad (2)$$

The entropy at the pump inlet is given by

$$S_S = f(T_S, P_S) \quad (3)$$

The hydrogen enthalpies at the turbopump inlet and discharge were determined from the functional relations:

$$h_S = f(P_S, T_S) \quad (4)$$

$$h_D = f(P_D, T_D) \quad (5)$$

Turbopump head rise. - The head rise across the turbopump was calculated from the following equation:

$$\Delta H = K \frac{g_c}{g} \left(\frac{P_D}{\rho_D} - \frac{P_S}{\rho_S} \right) \quad (6)$$

The hydrogen fluid properties used in the turbopump data analysis were obtained from reference 5.

RESULTS AND DISCUSSION

Turbopump Efficiency

The turbopump isentropic efficiency for tests 1, 2, and 3 is plotted against the discharge flow parameter Q_D/N in figure 8. The isentropic efficiency is the ratio of ideal enthalpy rise to the actual enthalpy rise (see eq. (1)). It does not include leakage or mechanical losses.

The pump tests were operated with inlet pressures of 35 and 50 psia (2.42×10^5 and 3.45×10^5 N/m²). The test results indicate, however, that the turbopump efficiency is es-

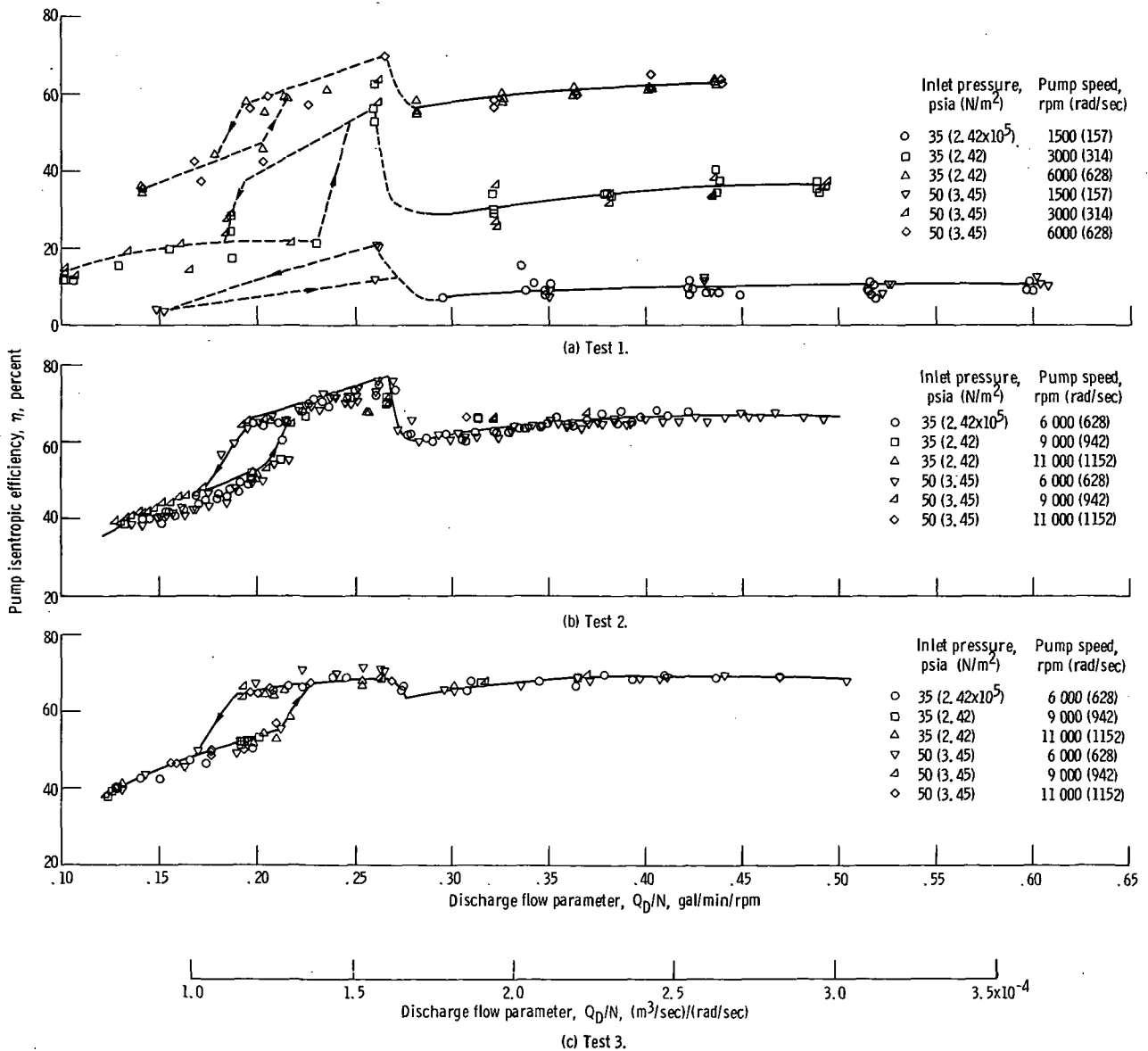


Figure 8. - Pump isentropic efficiency against discharge flow parameter for various pump speeds and inlet pressures.

TABLE II. - APPROXIMATE

ISENTROPIC EFFICIENCIES

Pump speed		Efficiency, percent
rpm	rad/sec	
1 500	157	9
3 000	314	32
6 000	628	63
9 000	942	66
11 000	1152	67

entially independent of the inlet pressure for the inlet pressures used.

At the rated discharge flow parameter Q_D/N of 0.346 gallon per minute per rpm ($2.09 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$), the approximate isentropic efficiencies for the five test speeds are given in table II. At speeds greater than about 6000 rpm (628 rad/sec), the efficiency appears to be insensitive to pump speed.

The increase in efficiency with speed (positive speed effect) is apparent in figure 8(a). This deviation from the pump affinity laws was expected in the low-speed operating regime.

A hysteresis trace is shown at the far left side of each curve in figure 8. The hysteresis is caused by the turbopump stall characteristics. With a decrease in flow rate, the pump enters stall at a Q_D/N of about 0.19 gallon per minute per rpm ($1.14 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$). And with increases in flow rate, the pump leaves the stall region at a Q_D/N of about 0.22 gallon per minute per rpm ($1.33 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$).

An unusual phenomenon appears in the efficiency curves of figure 8. As the flow rate is reduced below a Q_D/N value of about 0.28 gallon per minute per rpm ($1.69 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$), a step increase in efficiency occurs. An analysis of the pump temperature data showed that this abrupt increase in efficiency was caused by a sudden rise in the pump inlet fluid temperature, without a corresponding increase in the pump discharge fluid temperature. A plausible explanation for this is that fluid recirculation occurs in the pump impeller-inducer combination below a Q_D/N of about 0.28 gallon per minute per rpm ($1.69 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$). And the recirculated fluid carries some of the warmer pump fluid upstream to the temperature sensors. As a result, the actual enthalpy change $h_D - h_S$ across the pump appears to decrease, and the computed pump efficiency appears to increase. Analysis of the turbine data showed that no abrupt change in demand for power took place to correspond with the change in efficiency near a Q_D/N of 0.28 gallon per minute per rpm ($1.69 \times 10^{-4} \text{ (m}^3/\text{sec)}/(\text{rad/sec})$). Sufficient evidence was obtained from the test data to substantiate the recirculation theory. Therefore, it is concluded that the abnormal increase in efficiency at Q_D/N of about 0.28 gallon per minute per rpm

$(1.69 \times 10^{-4} \text{ (m}^3/\text{sec)})/(\text{rad/sec})$) is not a true pump characteristic. And the efficiency data in this region should not be used.

The calculated turbopump efficiencies presented in figure 8 were determined from the fluid temperature and pressure changes across the pump (see eqs. (1) to (5)).

The fluid temperature rise across the pump increased with pump speed, ranging from approximately 0.3° R (0.167 K) at 1500 rpm (157 rad/sec) to approximately 1.5° R (0.833 K) at 11 000 rpm (1152 rad/sec). As a result of these small temperature changes between the pump inlet and discharge (particularly at the low pump speeds), the computed turbopump efficiencies are greatly affected by small inaccuracies of the temperature sensors.

The effect of temperature measurement errors on efficiency has been estimated. For example, if the error in temperature rise across the pump was -0.1° R (-0.0556 K), a 1500-rpm (157-rad/sec) efficiency of 9 percent would change to about 11 percent, a 3000-rpm (314-rad/sec) efficiency of 32 percent would change to about 37 percent; and a 6000-rpm (628-rad/sec) efficiency of 63 percent would change to about 67 percent.

As stated in the APPARATUS AND EXPERIMENTAL PROCEDURE, the temperature sensors used in these tests were independently calibrated. The precision of the temperature measurements (with applied calibration corrections) was estimated to be within $\pm 0.08^\circ \text{ R}$ ($\pm 0.0445 \text{ K}$).

Errors in pressure measurements at the pump inlet and discharge would also affect the computed turbopump efficiencies. However, the effect of pressure measurement inaccuracies on efficiency is small compared to that of the temperature errors.

The pump isentropic efficiency is not a gross efficiency, and cannot be used to accurately calculate the input power requirements of the pump. The isentropic efficiency does not include mechanical losses and the loss of bearing coolant which leaks into the turbine. Since the pump and turbine share a common shaft, an external torque measuring device could not be used for these tests. The mechanical losses are probably less than 1 percent of the power input and can be neglected. The leakage loss to the turbine can be corrected for by multiplying the isentropic efficiency by the ratio of pump discharge flow to pump inlet flow. For the low-speed tests, this ratio averaged about 0.95 in the normal operating region. Hence, a reasonable estimate of the gross pump efficiency can be obtained by multiplying the isentropic efficiency by 0.95.

Turbopump Head Characteristics

Figure 9 shows the turbopump total head parameter $\Delta H/N^2$ as a function of pump discharge flow parameter Q_D/N for tests 1, 2, and 3. The presentation of data in speed-normalized fashion simplifies the comparison of runs at different speeds and shows

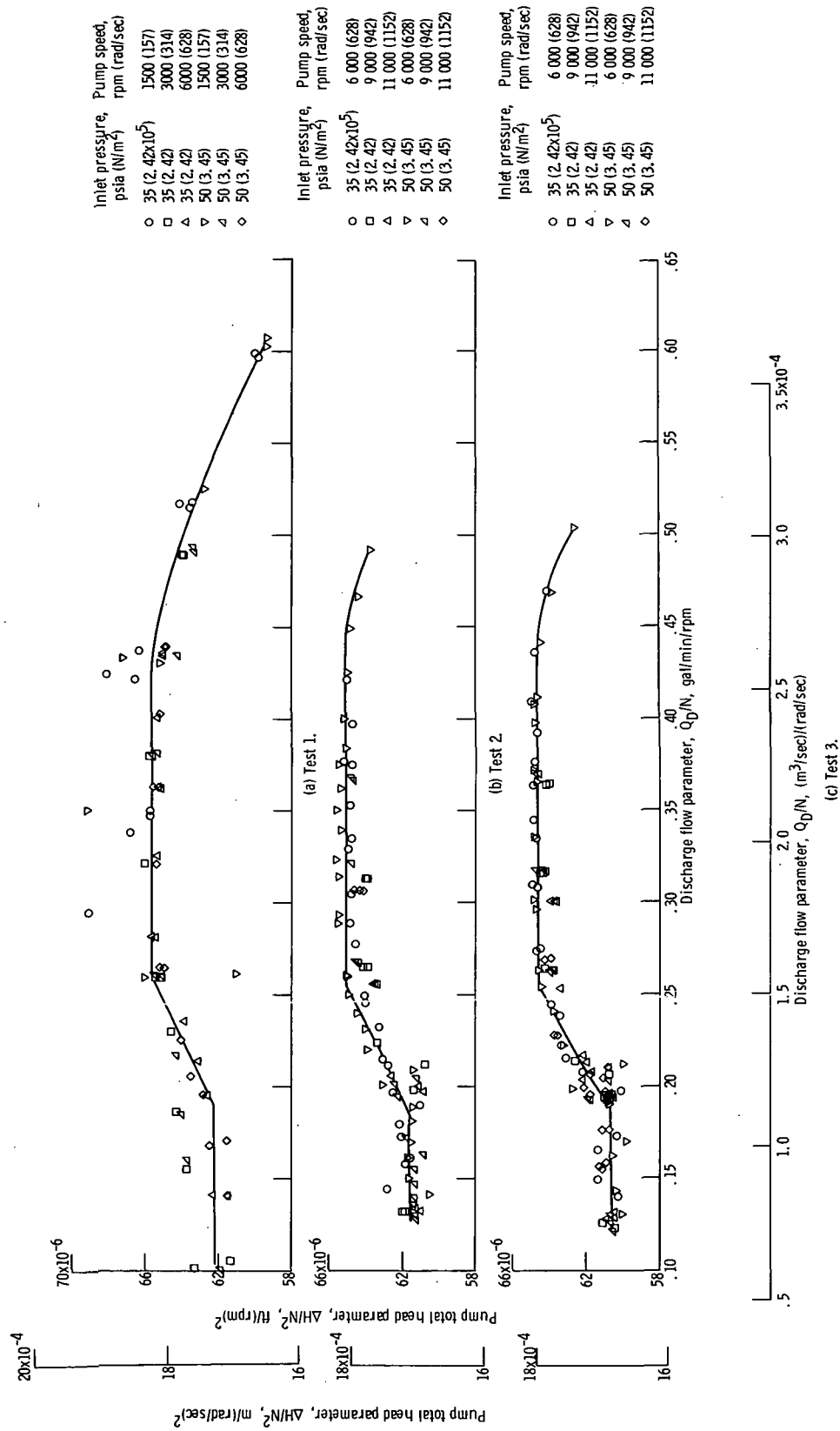


Figure 9. - Pump total head parameter against discharge flow parameter for various pump speeds and inlet pressures.

any deviation from the affinity laws. The normalized head curves for the three test runs (figs. 9(a) to (c)) are nearly identical.

In the normal operating region (i. e., for $Q_D/N > 0.26$ gal/min/rpm or 1.57×10^{-4} (m³/sec)/(rad/sec)), the head characteristic is relatively flat. This is typical of low-specific-speed centrifugal pumps.

At the rated discharge flow parameter Q_D/N of 0.346 gallon per minute per rpm (2.09×10^{-4} (m³/sec)/(rad/sec)), an average pump total head parameter $\Delta H/N^2$ of 65×10^{-6} feet per (rpm)² (18×10^{-4} m/(rad/sec)²) was obtained.

The dropoff in head at $Q_D/N > 0.44$ gallon per minute per rpm (2.65×10^{-4} (m³/sec)/(rad/sec)) appears to be the normal noncavitating head characteristics. If the dropoff was due to cavitation, the curves would not be in agreement for tests run at approximately the same net positive suction head (NPSH) but at different speeds, or for tests run at the same speed but at two different NPSH levels (35 and 50 psia (2.42×10^5 and 3.45×10^5 N/m²)).

As Q_D/N is reduced below 0.26 gallon per minute per rpm (1.57×10^{-4} (m³/sec)/(rad/sec)), the head decreases. The pump enters the stall region at a Q_D/N of about 0.19 gallon per minute per rpm (1.14×10^{-4} (m³/sec)/(rad/sec)) and leaves stall with increasing flow at a Q_D/N of about 0.22 gallon per minute per rpm (1.33×10^{-4} (m³/sec)/(rad/sec)).

Figure 10 shows the turbopump head characteristic in a flow-normalized form. The total head rise parameter $\Delta H/Q_D^2$ is plotted against the speed parameter N/Q_D . Data from tests 1, 2, and 3 at both 35 and 50 psia (2.42×10^5 and 3.45×10^5 N/m²) inlet pressure are included in figure 10. This figure shows the negative head region and the zero torque point.

The point at $N/Q_D = 0$ was obtained just before pump rotation started. Flow at this point is caused solely by the pressure in the hydrogen run tank. The point $\Delta H/Q_D^2 = 0$ was obtained during the speed transient at the beginning of each test. For a radial-vaned centrifugal pump, such as the Mark III - Model 4, the zero torque point occurs at zero speed. The normal operating range for the pump extends from a N/Q_D of about 2.0 to 3.8 rpm per gallon per minute (3.3×10^3 to 6.3×10^3 (rad/sec)/(m³/sec)). The stall region occurs at $N/Q_D > 4.5$ rpm per gallon per minute (7.5×10^3 (rad/sec)/(m³/sec)).

The head rise across the turbopump varies directly as the square of the pump speed. Consequently, at the low speeds, the head rise across the pump was relatively small. At the rated flow parameter of 0.346 gallon per minute per rpm (2.09×10^{-4} (m³/sec)/(rad/sec)), the head rise at 1500 rpm (157 rad/sec) was about 155 feet (47.3 m) of liquid hydrogen, or 4.4 psi (3.0×10^4 N/m²). As a result of the small head rise at low pump speeds, small errors in the measured pressures could have a significant effect on the total head parameter. The apparent scatter in the low-speed test data points of figure 9(a) may be due to small errors in the pressure measurements.

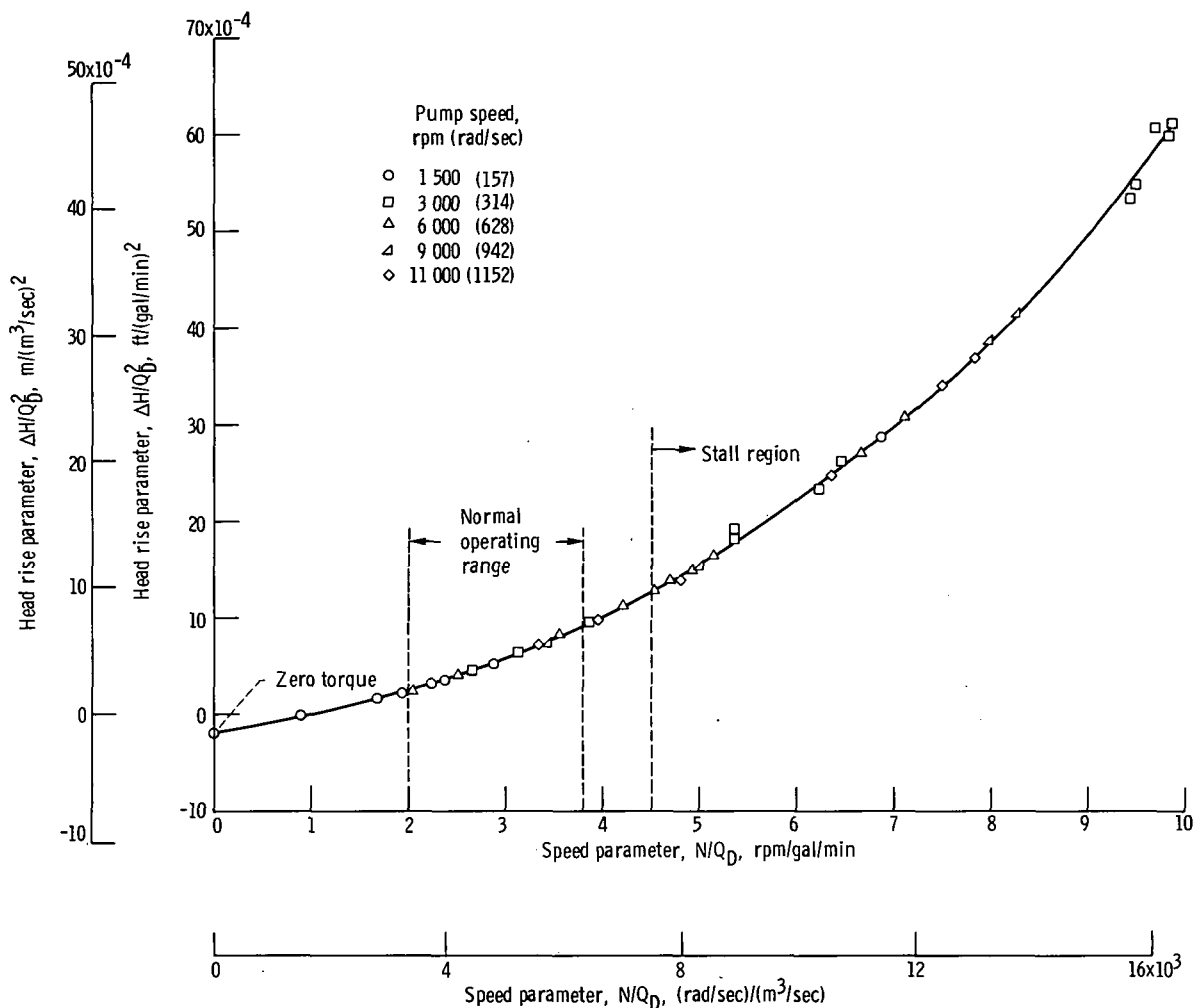


Figure 10. - Pump head rise parameter against speed parameter for speeds from 1500 to 11 000 rpm (157 to 1152 rad/sec).

CONCLUDING REMARKS

The turbopump data presented in this report cover a speed range from 6.7 to 49.0 percent of the rated pump speed. (Rated speed for this pump is 22 500 rpm (2356 rad/sec).) The major results from these pump tests are as follows:

1. The turbopump isentropic efficiency deviates noticeably from the affinity laws at speeds below about 6000 rpm (628 rad/sec). At the rated flow parameter Q_D/N of 0.346 gallon per minute per rpm (2.09×10^{-4} (m³/sec)/(rad/sec)), isentropic efficiencies of 9, 32, and 63 percent were obtained at speeds of 1500, 3000, and 6000 rpm (157, 314, and 628 rad/sec), respectively. At speeds greater than about 6000 rpm (628 rad/sec), the pump isentropic efficiency was nearly constant.

2. The turbopump head characteristics follow the affinity laws for the range of speeds investigated.

3. There were no obvious differences in the pump operating characteristics when the inlet pressure (NPSH) was changed from 35 psia ($2.42 \times 10^5 \text{ N/m}^2$) to 50 psia ($3.45 \times 10^5 \text{ N/m}^2$).

4. The abnormal increase noted in the isentropic efficiency when the discharge flow parameter Q_D/N was reduced below 0.28 gallon per minute per rpm ($1.69 \times 10^{-4} (\text{m}^3/\text{sec})/(\text{rad}/\text{sec})$) is not a true pump characteristic. Therefore, the efficiency data in this region should not be used. Analysis of the test data indicates that fluid recirculation occurs in the pump impeller-inducer combination below a Q_D/N of about 0.28 gallon per minute per rpm ($1.69 \times 10^{-4} (\text{m}^3/\text{sec})/(\text{rad}/\text{sec})$). The recirculated fluid causes erroneously high measured inlet temperatures. And these, in turn, result in a higher calculated (but untrue) isentropic efficiency.

5. There was no noticeable shift in the pump stall line when the pump speed was reduced.

Lewis Research Center,

National Aeronautics and Space Administration

Cleveland, Ohio, May 28, 1969,

120-27-04-82-22.

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